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FINAL REPORT

PRELIMINARY DESIGN OF THE CRYOGENIC COOLING SYSTEM FOR THE CRYOGENIC COOLED LIMB SCANNING INTERFEROMETER RADIOMETER (CLIR) - PHASE II

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1.0 INTRODUCTION

This study is the second phase of a preliminary design for the cryogenic cooling system for the Cryogenic Cooled Limb Scanning Interferometer Radiometer (CLIR) instrument to be flown on the Atmospheric-Magnetospheric Physics Satellite (AMPS).

The Phase I study was completed on May 31, 1978. The Phase I Study included extensive trades comparing the relative effectiveness of various stored cryogen systems, including solid cryogens and solid cryogens in combination with helium. Comparisons were made between single, dual and triple stages.

The results(1) indicated that the single stage approach in combination with vapor cooling of the optics (30°K) and radiation baffle (110°K) offered advantages over dual stage approaches. The primary candidates that evolved from this effort were solid hydrogen and supercritical helium. Although solid hydrogen was a superior choice in terms of size, weight, and temperature stability the supercritical helium was selected for reasons of safety and more current state of development.

The resulting phase I design consisted of a cylindrical helium tank with hemispherical domes with a length somewhat over two meters.

The permissible envelope for the study was consistent with a center of gravity type pointing system (SIPS) and as such placed principal limitations on the diameter of the package with less emphasis on the length.

The Phase II study presented here called for a re-configuration of the cryogenic instrument package for operation with the ASPS pointing system, which is an end mounted type and as such allows larger diameters but may benefit from or require a reduced package length.

The Phase II Study was directed at the re-configuration of the cryogenic system for this mounting system and included refinements of the analysis in both thermal and structural areas.

The instrument requirements for temperature and vent gas cooling flow rates were assumed to be the same as in the Phase I study since no additional or new instrument development work was done during the Phase II study period.

The principal restraint of the system was an overall package length of 2 meters maximum which included the 1 meter high instrument. Within this restraint several configurations were examined, the primary ones being a toroidal helium tank with the instrument inside and a spheroidal tank with the instrument mounted on the top.

The comparisons showed the spheroidal tank to be the best choice and more detailed studies were performed on that configuration.

2.0 INSTRUMENT COOLING REQUIREMENTS

The instrument cooling requirements used in this study were the same as used in the Phase I effort. The criteria utilized are summarized below.

	Allowable Temperature, °K	Heat Load, W
Detectors	7-12	0.065
Optics Module	30	0.81
Baffles	110	13.28

Prior studies of vapor cooling of the instrument in Phase I indicated that a helium flow rate to the instrument of 0.2 lbs/hr will meet these requirements.

For this flow rate condition, the baffle sets the requirement for vetn gas flow rate, and is cooled to 110°K while the resulting optics module temperature is 24°K for this flow rate.

The detailed break-down of the instrument heat rates and the variation in instrument temperature with variations in flow rate have been calculated and presented in the Phase I effort.

The mission lifetime was 30 days.

3.0 CONFIGURATION SELECTION

One of the primary design constraints for the study was that the overall cooler/instrument length was not to exceed 200 cm. Consideration of alternate approaches was largely driven by this criteria.

Prior to selection of the baseline cooler configuration (Section 4) several alternative options were studied. The three primary options were the toroidal tank, ellipsoidal tank with instrument mounted on the side and the selected baseline of a spheroidal tank with instrument mounted above. Each of the three configurations as well as the baseline cooler developed in the phase I study are shown schematically in Figure 3-1. Shown in Table 3-1 is a summary of the major features exhibited by each design.

The toroidal design offered a compact integration between the experiment and cooler. By placing the experiment package between the inner walls of the toroid, a maximum 169.7 cm diameter by 149.5 cm long volume envelope could be realized. The primary problem with this design is the relatively large mass of the cooler system which is 412 kg; twice the phase I baseline cooler mass. The main reason for the increased system mass is the need for a larger and, in some locations, a much stiffer vacuum shell.

In routing of the cooler vent line from the cooler to the instrument focal plane, penetration into a cavity having an ambient temperature boundary could not be avoided. Although it was felt that thermal isolation techniques could be employed (insulation, shielding, etc.) to reduce the heat leak to an acceptable level, other system configuration approaches were considered. These thermal problems along with the high mass ultimately led to the selection of an alternate design.

Attention was shifted toward an ellipsoidal tank configuration having a more direct coupling to the instrument in the cold cavity. Of the two options shown in Figure 3-1, the side mounted experiment configuration was found to require a more complex gimbal support configuration than the top mounted con-

FIG. 3-I SCHEMATIC SUMMARY OF SYSTEMS STUDIES

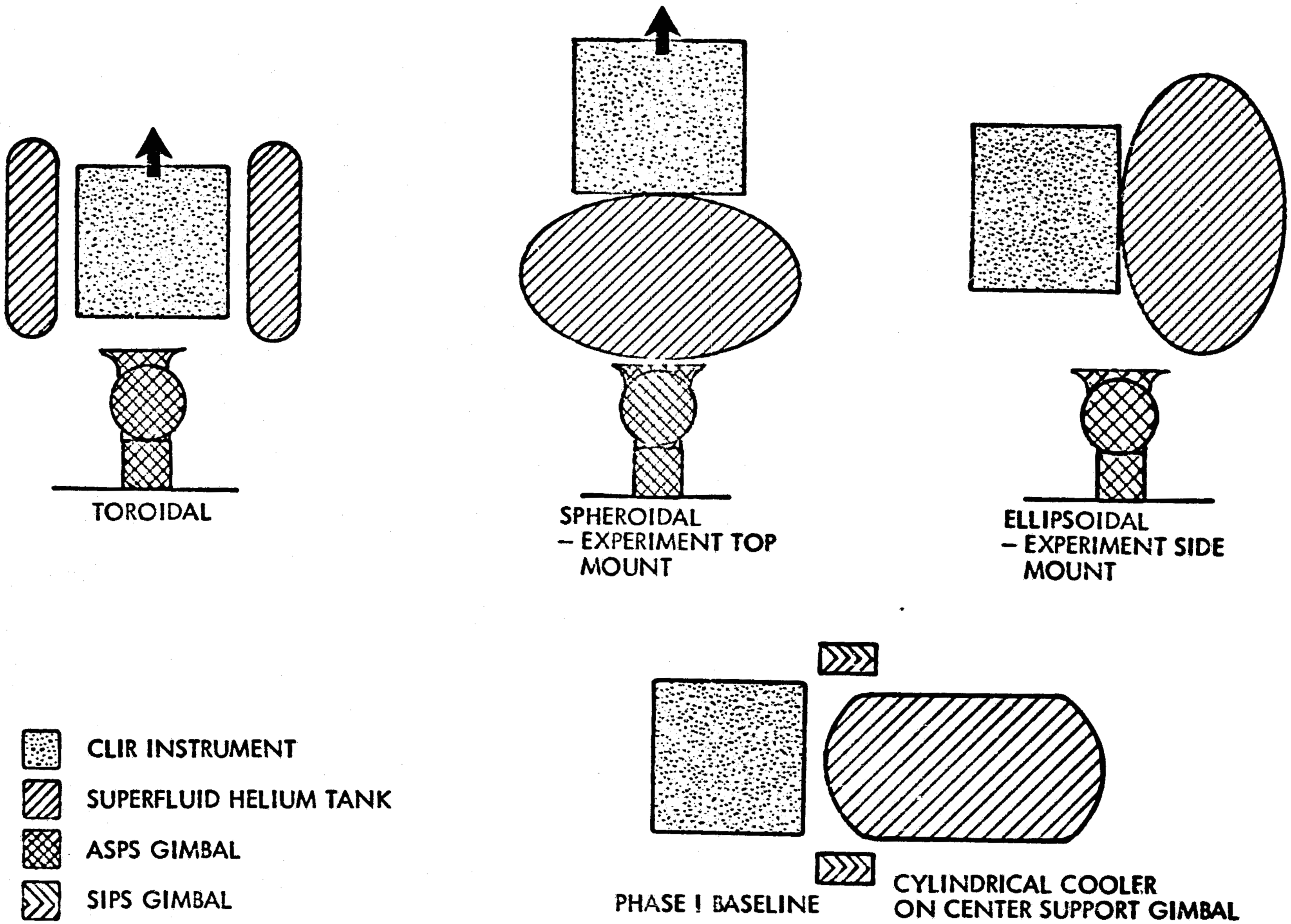


TABLE 3-I MAJOR FEATURES OF SYSTEMS STUDIED

CONFIGURATION	TOTAL LOADED MASS (kg)	STRUCTURAL CONSIDERATION	VOLUME ENVELOPE (INSTRUMENT + COOLER)	THERMAL CONSIDERATION
PHASE I BASELINE; CYLINDRICAL TANK, INSTRUMENT ABOVE C.G. POINTING SYSTEM (SIPS)	197.1 kg	• PRIMARY RESONANCE = 28 Hz	300 cm LENGTH 95.5 cm DIA.	• Q_{DEWAR} = 0.415 W
PHASE II CANDIDATES END MOUNTED POINTING SYSTEM (ASPS) • TOROIDAL	412.0 kg	• NOT ANALYZED, BUT NO PROBLEMS EXPECTED	149.5 cm LENGTH 169.7 cm DIA.	• VENT LINE EXPOSED TO AMBIENT BOUNDARY TEMPERATURES
• ELLIPSOID/ INSTRUMENT SIDE-MOUNT	NOT DETERMINED	• DIFFICULT TO ANALYZE BECAUSE OF NON-RADIAL SYMMETRY	NOT DETERMINED	NOT DETERMINED
• SPHEROID/ INSTRUMENT TOP-MOUNT	339.9 kg	• PRIMARY RESONANCE = 21.5 Hz • SUPPORT CONFIGURATION EASIER TO IMPLEMENT THAN SIDE-MOUNT CONFIGURATION	200 cm LENGTH 152.4 cm DIA.	• Q_{DEWAR} = 0.522 W • CONTINUOUS VENT GAS COOLING OF SUPPORT TUBE

figuration, primarily because of the lack of radial symmetry. This lack of symmetry also leads to a very difficult structural analysis. Further, no clear thermal advantage over the top mounted approach would be gained as a similar cooler/experiment interface with the top mounted configuration would be employed. For these reasons, the top mounted configuration was selected for further analysis.

The third approach consisted of a spheroidal tank with the instrument mounted on the top. Allowing for vacuum shell, experiment door, multilayer, and support structure, the CLIR experiment was located as close to the end of the 200 cm limit as possible. Within the volume remaining, a maximum cooler semi-minor axis (tank height) was determined (38.1 cm) which also allows for insulation around the cooler. The remaining semiaxis was then selected such as to make a spheroid (to preserve radial symmetry) having a total volume of 712 liters. Adding the thicknesses required for multilayer on the tank sides the total volume envelope is 152.4 cm dia by 200 cm long. The mass of the cooler is considerably less than that of the toroidal configuration at 339.9 kg however, this still represents a 70% increase in mass over the phase I baseline cooler mass. Again, as with the toroidal design, this increased mass is mostly made up by a larger and stiffer vacuum shell over that required for the SIPS gimbal.

This arrangement of cooler and experiment provides for relatively easy access of the vent cooling gas line to the experiment through a low temperature region of the package. This system appears to be a simpler approach than the other two considered which leads to reduced lost and schedule relative to the other approaches.

4.0 BASELINE SYSTEM

4.1 SUMMARY

Fig. 4-1 summarizes some of the major thermal, mechanical, and operating considerations for the CLIR cooler system. As with the cooler designed in the Phase I study, cooling of the instrument will be provided by the cold vent gas from helium maintained in a supercritical state between 6 - 12°K. The tank internal pressure will be maintained constant at 4.1×10^5 nt/m^2 (60 psia) during withdrawal by an absolute pressure relief valve located at the exit of the tank vent line. The cold vent gas from the dewar is routed directly to the focal plane heat exchangers before being used to cool either the instruments optical bench or radiation baffle to 30°K and 104°K respectively. Upon exit from the experiment, the gas is used to cool first a radiation shield and second, a portion of the fiberglas support tube.

For the initial fill, the tank is filled to a ten per cent (10%) ullage with normal boiling point helium. The fill and vent lines are then closed and the system is allowed to self-pressurize to 4.1×10^5 nt/m^2 (60 psia) after a minimum 68 hour ground hold time. This fill procedure will simplify the ground support equipment required when compared to filling initially with supercritical helium at high pressure.

After venting begins, the expected steady state parasitic heat load to the dewar will be 522 mw with a vapor cooled shield temperature of 114.2°K. In order to maintain the optical elements at 30°K, a flow rate of 2.5×10^{-2} gm/sec (0.2 lb/hr) is required. Depending on the cryogen temperature, between 15 to 991 mw of auxiliary electrical heater power will be necessary in order to maintain this flow rate. Because the power required is a function of the helium temperature in the dewar, a feedback control loop is required to provide the proper experiment temperature. This feedback control loop will sense the instrument temperatures and vary the power into an electrical resistance heater wrapped about the dewar, adjusting the flow rate as necessary.

FIG. 4-1 SUMMARY OF PRIMARY PARAMETERS OF BASELINE SYSTEM

MECHANICAL

- OVERALL DIAMETER (INCL. VACUUM SHELL) 152.4 cm
- CRYOGEN TANK VOLUME 712 LITER
- DRY WEIGHT (INCL. 20% MARGIN) 259.9 kg
- USABLE HELIUM WEIGHT 68.1 kg
- RESIDUAL HELIUM WEIGHT 11.9 kg
- TOTAL LOADED WEIGHT 339.9 kg
- PRIMARY RESONANCE 21.5 Hz
- DESIGN SAFETY CRITERIA
  - SUPPORT TUBE — SURVIVE QUAL RANDOM WITH 3σ PROBABILITY
  - HELIUM TANK — SAFETY FACTOR = 4 (BURST)
  - VACUUM SHELL — SAFETY FACTOR = 2 (BUCKLING)

THERMAL

- HEAT LOAD TO HELIUM = 0.522 W
  - CONTINUOUS VENT GAS COOLING OF SUPPORT TUBE BETWEEN AMBIENT AND VAPOR COOLED SHIELD
  - SINGLE VAPOR-COOLED SHIELD (T = 114.2 K)
  - 6.4 cm SILK NET/DOUBLE ALUMINIZED MYLAR INSULATION

OPERATING CONDITIONS

- SUPERCRITICAL HELIUM CRYOGEN
- CONSTANT PRESSURE WITHDRAWAL AT 4.1×10^5 nt/m^2 (60 psia)
- REQUIRED FLOW RATE TO INSTRUMENT 2.5×10^{-2} gm/s (0.2 lb/hr)
- REQUIRED HEAT INPUT FOR WITHDRAWAL OF 2.5×10^{-2} gm/s HELIUM — .02 W @ 6 K; .99 W @ 12 K
- FILLING MODE
  - FILL TO 90% (MIN) WITH NBP HELIUM
  - SELF PRESSURIZE TO 4.1×10^5 nt/m^2 (60 psia)
- GROUND HOLD TIME (FROM END OF FILL TO START OF VENT) 68 hr

Unlike the cooler developed in Phase I of this program, the present cooler configuration makes use of the vent gas from the radiation shield to cool that portion of the support tube between ambient and the radiation shield. The need for vapor cooling of the support tube in this configuration was made necessary because of packaging constraints which required the use of a shorter yet thicker (larger radii) support tube.

Overall length of the cooler from the ASPS interface to the cooler/instrument interface is 140.7 cm. By recessing the experiment into the cooler cavity, ample space is provided to insure the maximum length envelope of 200 cm will not be exceeded.

4.2 COOLER CONSTRUCTION

A layout showing hardware detail of the CLIR cooler is shown in Fig. 4-2.

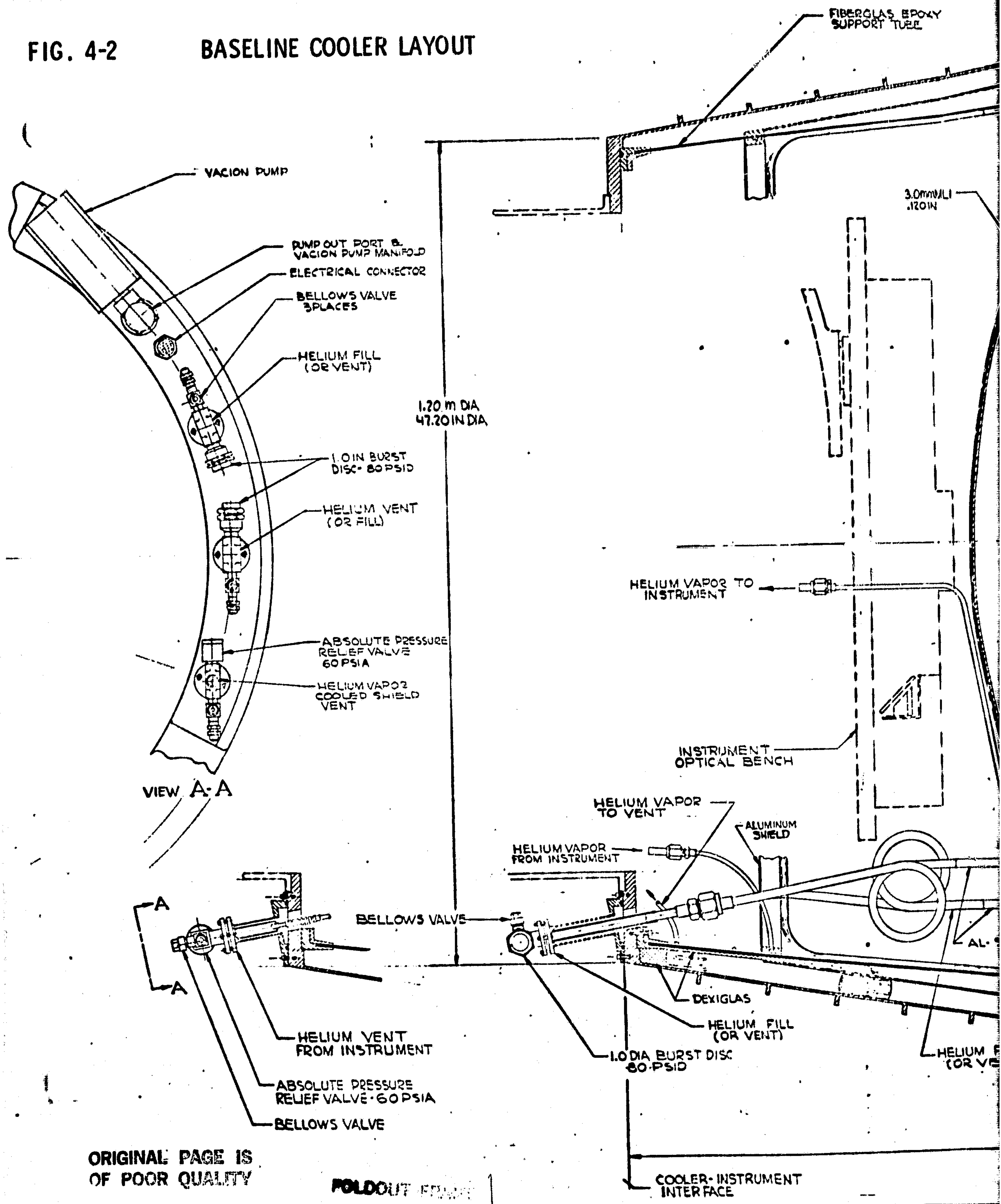
Helium Tank

The helium tank is a 712 liter 6061 T4 aluminum oblate spheroid having major and minor semiaxes of 66.79 cm and 38.10 cm respectively. The tank which has a uniform thickness of 0.318 cm is spun formed in two halves and joined along the major axis by a weld joint. Three penetrations are made into the front portion of the tank where the two fill/vent lines and a single fluid withdrawal line is mated to the dewar, also by a weld bond. All welded joints will also incorporate an epoxy bonded element to provide additional assurance against vacuum leaks.

Support Tube

Both the dewar and vapor cooled radiation shield are supported by a single fiberglas support tube. The support tube is mounted to a strongback portion of the vacuum shell so that the launch and return structural loads will be transmitted to the ASPS gimbal mounting pad through the vacuum shell. The previous Phase I design provided support directly to the gimbal interface. Although

FIG. 4-2 BASELINE COOLER LAYOUT



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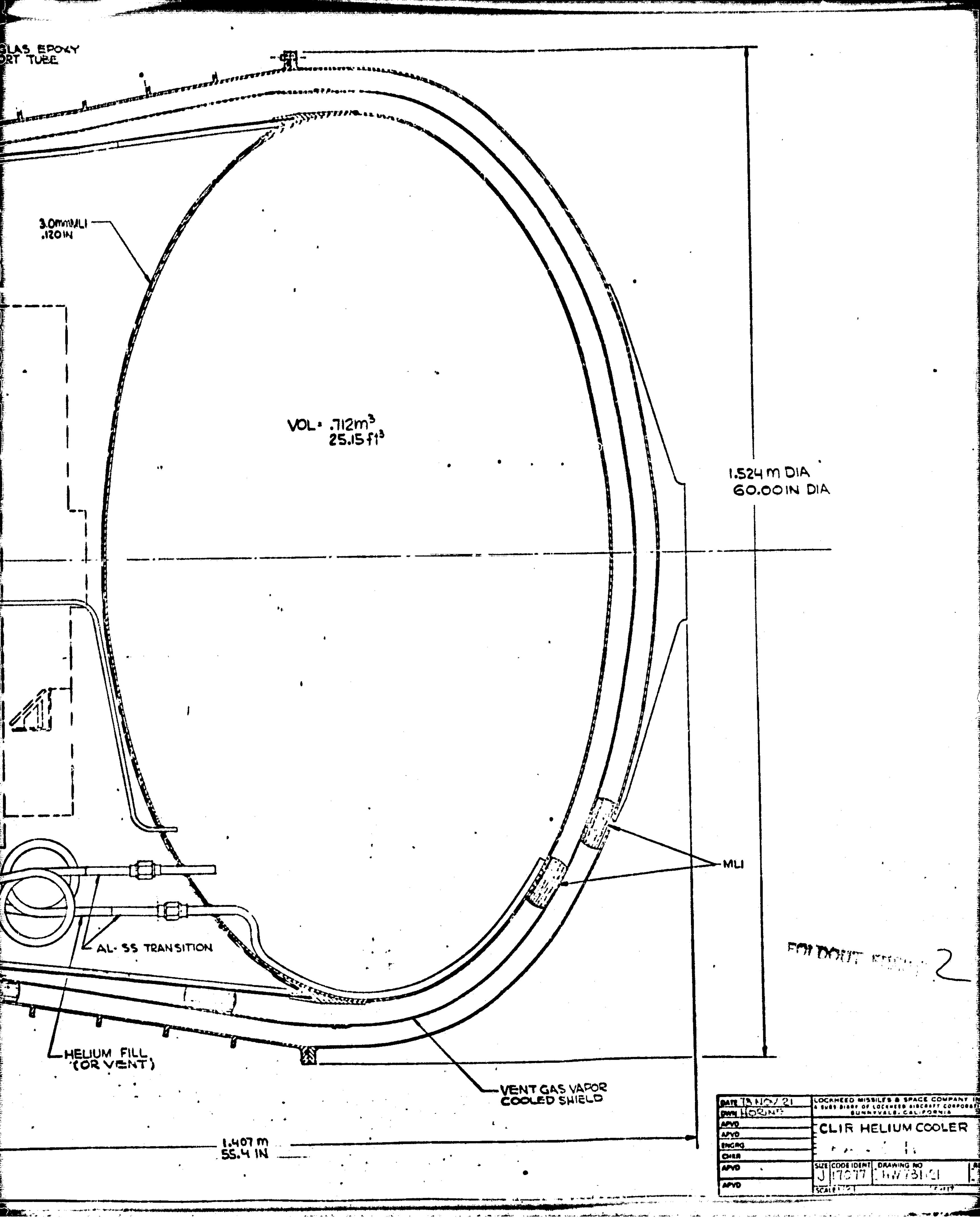
FOLDOUT FRAME 1



LAS EPOXY
RT TUBE
3.0mm MLI
.120 IN
VOL = .712 m³ 25.15 ft³
1.524 m DIA
60.00 IN DIA
MLI
AL-SS TRANSITION
HELIUM FILL
(OR VENT)
VENT GAS VAPOR
COOLED SHIELD
1.407 m
55.4 IN
LOCKHEED MISSILES & SPACE COMPANY
SUNNYVALE, CALIFORNIA
CLIR HELIUM COOLER
DATE
DWN
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DRAWING NO
SCALE
2

this is a more desirable situation, structural analyses (see Section 4.5) on the vacuum shell show that design criteria for the shell to survive buckling will make for a sufficiently "stiff enough" structural member to insure that the difference between the two mounting configurations will be negligible with respect to the structural loads to be incurred by the cooler.

The tube itself is a monocoque 0.203 cm thick inverted right circular truncated cone constructed of 1543/E787 fiberglas epoxy. The minimum (at the strongback) and maximum tube radii is 57.71 cm and 65.94 cm respectively, and has an unsupported slant length of 73.56 cm. The support tube flanges are each machined in two pieces so as to contain the support tube in a clam-shell fashion when assembled. Attachment to the strongback is made by mechanical fasteners while a more permanent epoxy bond is made to the dewar.

Vapor-Cooled Shield

A vapor cooled shield is provided for thermal isolation about the rear portion of the cooler and the cold section of the fiberglas support tube. Around the cooler, the shield follows the contour of the dewar, maintaining a uniform 3.2 cm spacing. As the shield extends over the fiberglas support tube, it also takes the shape of a right circular truncated cone, only having a slightly steeper slope. The steeper slope causes the shield to intersect the support tube with a gentile transition at a point 57.15 cm from the dewar. At the support tube, the shield is epoxy bonded to the tube along a 2.92 cm wide annular pad. To cool the shield, a similar annular pad is epoxy bonded to the inner wall of the tube at the same location as the outer pad. The vent gas plumbing is routed to the inner ring where contact is made over a full circumference. An aluminum shield is also coupled to the inner annulus to act as a radiation shield for the cold components within the main vacuum cavity. Exact termination of this shield is as yet undefined, pending further definition of the experimentor's package.

Vacuum Shell

The vacuum shell is cantilevered off the ASPS gimbal mounting system which in turn provides the structural support for both the cooler and the CLIR experiment. The entire shell is a two piece construction of 6061-T6 aluminum having a common O-ring vacuum seal.

The half spheroidal dome piece is 0.381 cm thick, reinforced in the center by six 2.5 cm thick radial spider arms. A flange is provided for connection to the truncated cone/strongback piece. The truncated cone section is an 0.381 cm thick, ring-stiffened section having a slope parallel to that of the vapor cooled shield. The strongback portion of the piece is a 1.91 cm thick annulus which provides a structural base for both the cooler and experiment. This strongback represents the cooler/experiment interface, as well as the mounting surface for all cooler plumbing lines.

Insulation

A 3.2 cm thick blanket of silk net/double aluminized mylar insulation completely surrounds both the vapor cooled radiation shield and the helium dewar.

As indicated above, the vacuum shell along the conical section is parallel to the radiation shield so that between these two members, the layer density is constant at 14.6 layers/cm. In the region beyond the radiation shield, over the fiberglas support tube, the number of layers remains constant. However, the change in slope between the radiation shield and the support tubes forces a slight bit of crushing, increasing the layer density to a maximum of 16.8 layers/cm. Dexiglas is inserted at the edge of the blanket to aid in reducing edge effect heat leaks.

In the region between the vapor-cooled shield and the fiberglas support tube, the layer density will remain constant throughout the length of the shield at 14.6 layers/cm, the same density as over the dewar back section. To

achieve this, each layer as it extends over from the back surface of the dewar will be individually "cut-to-length" at the point where contact is made with the fiberglas support tube. By varying the length of each wrapping, the MLI edge temperature will more nearly match the support tube temperature as inner layers are coupled to the colder portion of the tube. This has the result of nearly eliminating any major edge effect heat leaks. To prevent the cooler cavity from being exposed to the support tube wall gradient, the inner surface of the tube and radiation shield is insulated with a 0.64 cm thick blanket. To implement this wrapping, the required contour is molded from a thin mylar sheet and the insultion is wrapped over the mold. After being wrapped, the entire blanket/mylar assembly is positioned inside the tube. Finally, the front surface of the helium dewar is wrapped with a 0.32 cm thick blanket of silk net double-aluminized mylar.

Plumbing

Shown in Fig. 4-2, in view A-A is the plumbing interface with the cooler. Mounted in the strongback portion of the vacuum shell is an annular sector plumbing access flange which contains the cooler fill/vent line, vapor cooled shield vent line, vacuum pump-out access (including vac-ion pump), and electrical feedthrough. The pumpout line to the insultion space consists of a 2.5 cm diameter Cryolab valve to provide access to an auxiliary sorption pump system and an 8 ℓ/s vacion pump to ultimately maintain pressures in the insulation space at less than 10^{-5} torr. The electrical feedthru provides access to tank thermometry, several redundant strain gauges used to monitor the tank internal pressure, and a heater located on the tank to control the cryogen venting rate.

Three plumbing lines comprise the helium fill and vent lines; all being 1.3 cm in diameter. In the dewar, a short aluminum section is welded to the tank wall. Internal to the tank, these lines terminate such that filling of the tank may be made to within a six per cent (6%) ullage in either a vertical or horizontal tank position. This section couples to an aluminum/stainless transition tube, keeping the length of aluminum required to a minimum.

Within the stainless section, several loops are made in the line to first eliminate potentially harmful thermal stresses and to second reduce the plumbing heat leak to the helium dewar. A second connection is made near the strongback to allow for easy removal of the plumbing access flange. All couplings utilize "Gamah" type fittings or equivalent. On the outside of the tank, the lines are supported by a bayonet fitting capped by a 0.95 cm diameter Nupro metal bellows type access valve in parallel with a 2.5 cm diameter, 5.5×10^5 nt/m^2 (80 lb/in^2) differential burst disk. This level is set above the 4.1×10^5 nt/m^2 (60 lb/in^2) absolute pressure of the tank to insure no venting of the cryogen occurs through these lines. Instead, venting of the cryogen is accomplished through the vapor flow vent line.

A schematic of the plumbing is shown in Fig. 4-3. The optics and baffle temperatures are monitored and maintained constant through a cryogen tank heater feedback control loop to control helium vent rate. Supercritical helium is vented through the two detector focal planes maintaining temperatures to less than 13°K. The cold vapor continues through the instrument, cooling the optics box to 30°K and radiation baffle to 104°K. From the radiation baffle, the cooled vapor is directed to the cooler vapor cooled shield which is driven to 114.2°K. The gas is then used to vapor cool the support tube, wrapping one complete turn in helical fashion along the tube length. At room temperature, the gas is finally exited from the cooler through a 4.1×10^5 nt/m^2 (60 lb/in^2) absolute pressure relief valve which controls the pressure of the tank. In parallel with the pressure relief valve is a Nupro metal bellows-type valve which can be used to blow the tank down to one atmosphere pressure for refill operation on ground.

Mass

A mass summary of the main components required for the CLIR cooler is shown in Fig. 4-4. A total cooler system mass of 340 Kg (748 lbs) is predicted for the supercritical helium cooler, of which 80 Kg (176 lbs) or 24 per cent is due to the helium cryogen.

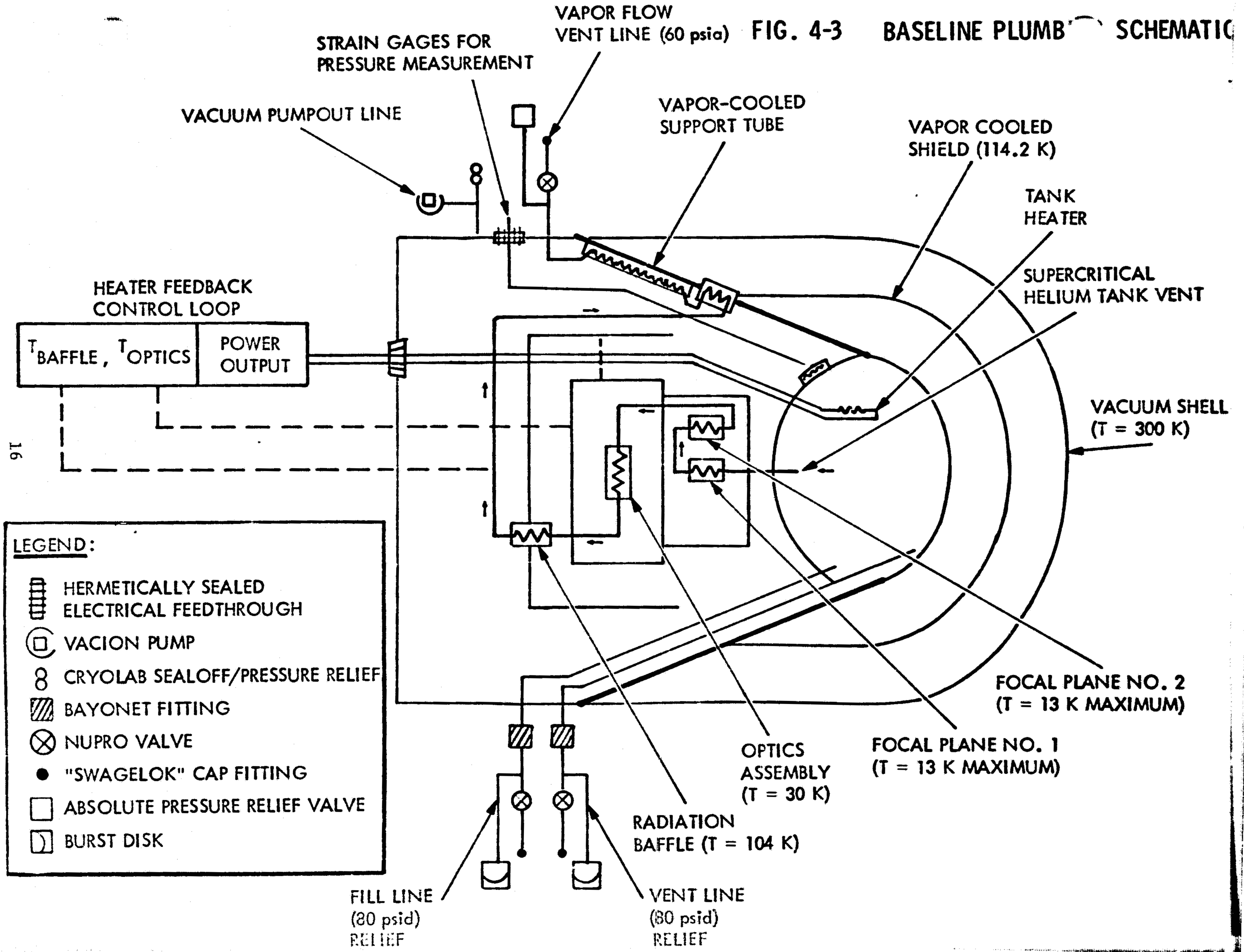


FIG. 4-3 BASELINE PLUMB[illegible] SCHEMATIC

FIG. 4-4 COOLER WEIGHT SUMMARY

ITEM	MATERIAL	THICKNESS (cm)	MASS (kg/lb)
PRIMARY CRYOGEN TANK	ALUMINUM	0.381	41.5/91.7
VAPOR-COOLED SHIELD	ALUMINUM	0.076	11.6/25.5
VACUUM SHELL	ALUMINUM		123/271
SUPPORT TUBE	FIBERGLAS	0.203	10.8/23.7
MOUNTING FLANGES	ALUMINUM		16.1/35.4
MLI			5.8/12.7
PLUMBING			2.7/6.0
VACION PUMP (INCLUDING MAGNETS)			4.1/9.0
MISCELLANEOUS HARDWARE			1.0/2.2
TOTAL DRY MASS			216/476
20% MARGIN			43.3/95.2
DRY MASS WITH MARGIN			259.9/571.2
CRYOGEN MASS			80.0/176.3
TOTAL LOADED MASS			339.9/747.6

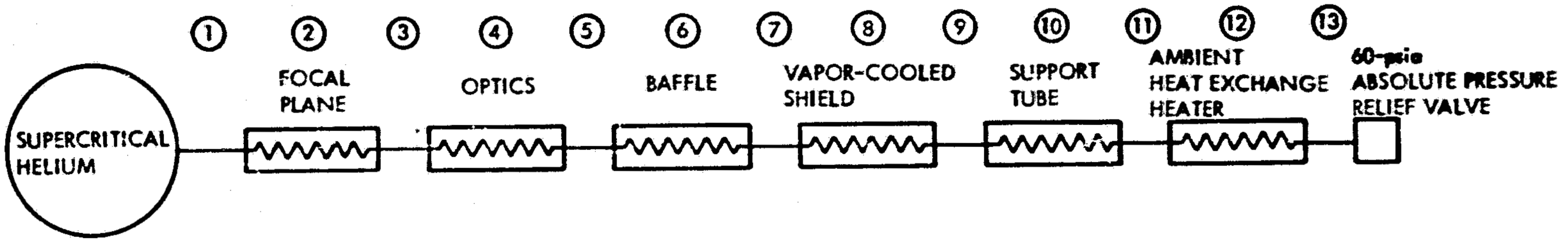
Comparing to the cooler developed in the Phase I initial study, the total system mass has increased approximately 143 Kg. The main reason for this increase is in the mass of the vacuum shell. In the current design, extra stiffening is required first at the base of the cooler where coupling to the ASPS gimbal system occurs, and second at the strongback. Second order increases were observed in the vapor-cooled shield and mounting flanges. It is felt that some of this increase in mass will be recovered when the total system mass is determined. Since the experiment is recessed into the cooler vacuum space, a smaller vacuum cover will be required for the experiment than that required in the Phase I Study.

4.3 VENT LINE ANALYSIS

Fig. 4-5 shows a schematic of the vent line in which the different heat exchanger sections and connecting line configurations are described. The lines between the heat exchanger sections are assumed to be isothermal at the temperature of the component from which the gas flows. Number and types of bends in the experiment section are approximate pending more experiment definition. However, changes will have a negligible effect on the total pressure drop. Selection of the line lengths in the isothermal sections was taken directly from the cooler layout for sections 7, 9, 11 and 13. For those sections within the experiment, a conservative extimate was made based on the small details available on the experiment cavity.

Each focal plane heat exchanger (shown lumped in Fig. 4-5) was assumed to be comprised of 33 layers of 200 mesh screen, layered in a 1.24 cm diameter tube uniformly spread along a 2.54 cm length. Using the friction factor and pressure drop relationship suggested by Kays and London[1] for flow through a woven screen cloth, the pressure drop through each focal plane heat exchanger is found to be only 19.5 nt/m^2 or 39.1 nt/m^2 (0.0057 psia) for the entire focal plane heat exchanger section.

FIG. 4-5 VENT LINE SCHEMATIC



ISOTHERMAL LINE CONFIGURATIONS

LINE	LENGTH (cm)	NO. OF 45-deg BENDS	NO. OF 90-deg BENDS	NO. OF 180-deg BENDS	TEMPERATURE (K)
1	60		4		6.0
3	25		3		12.0
5	20		3		30.0
7	140	2	2		104.0
9	19			1	114.2
11	11	2			300.0
13	11				300.0

HEAT EXCHANGER LINE CONFIGURATION

LINE	LENGTH (cm)
2	N/A – 2 EA WIRE MESH SCREENS
4	306
6	146
8	433
10	143
12	55

The remaining heat exchanger sections utilize either continuous (along the support tube) or single point (at all other locations) vent gas cooling. Solutions to the Graetz problem[2] shows that 95% of the maximum heat removing capacity of a venting gas is achieved provided the dimensionless flow parameter $\dot{m}c_{p/\pi kL}$ is less than 1.2. The required length in each heat exchange section was determined from this relationship and then doubled for conservatism. Each value was then compared to that found in the layout. In all cases except the optics box heat exchanger the mechanical requirement exceeded the thermal requirement. The largest value of the two is as shown in Fig. 4-5. For the vent line cooling of the support tube, the length required for one helical revolution about the entire tube slant length is used.

At a mass flow rate of 2.5×10^{-2} gm/sec, the Reynolds number in a 0.635 cm diameter tube can be approximated for helium by $Re = 10415/T^{0.656}$ so that between 30 - 300°K, the Reynolds number varies between 1100 and 250 respectively, making the flow laminar. Using the classical relationship for fully developed flow

$$\Delta p = 4\ (L/D)\ (\frac{\rho U^2}{2})\ f$$

where $f = 16/Re$.

The pressure drop per length of line at 4.14×10^{-5} nt/m^2 (60 psia) and mass flow rate of 2.5×10^{-2} gm/sec is given by

$$\Delta p/L = 1.515\times10^{-5} T^{1.656}\ \text{nt/m}^2\text{/cm}$$

(the $T^{1.656}$ factor occurs because of temperature dependent helium viscosity and the density relationships used in evaluating the Reynolds number). As can be seen, the maximum pressure gradient occurs at the higher temperature portion of the line. As an upper bound, if the entire line were assumed to be at ambient temperature (300°K) the pressure drop realized would be 325 nt/m^2 (0.05 psia) where an equivalent length including bends of 1700 cm was assumed. As actual temperature within the vent line are less than 100°K through 88% of the line, it can be assumed that flow within the vent line will be isobaric.

4.4 THERMAL ANALYSIS

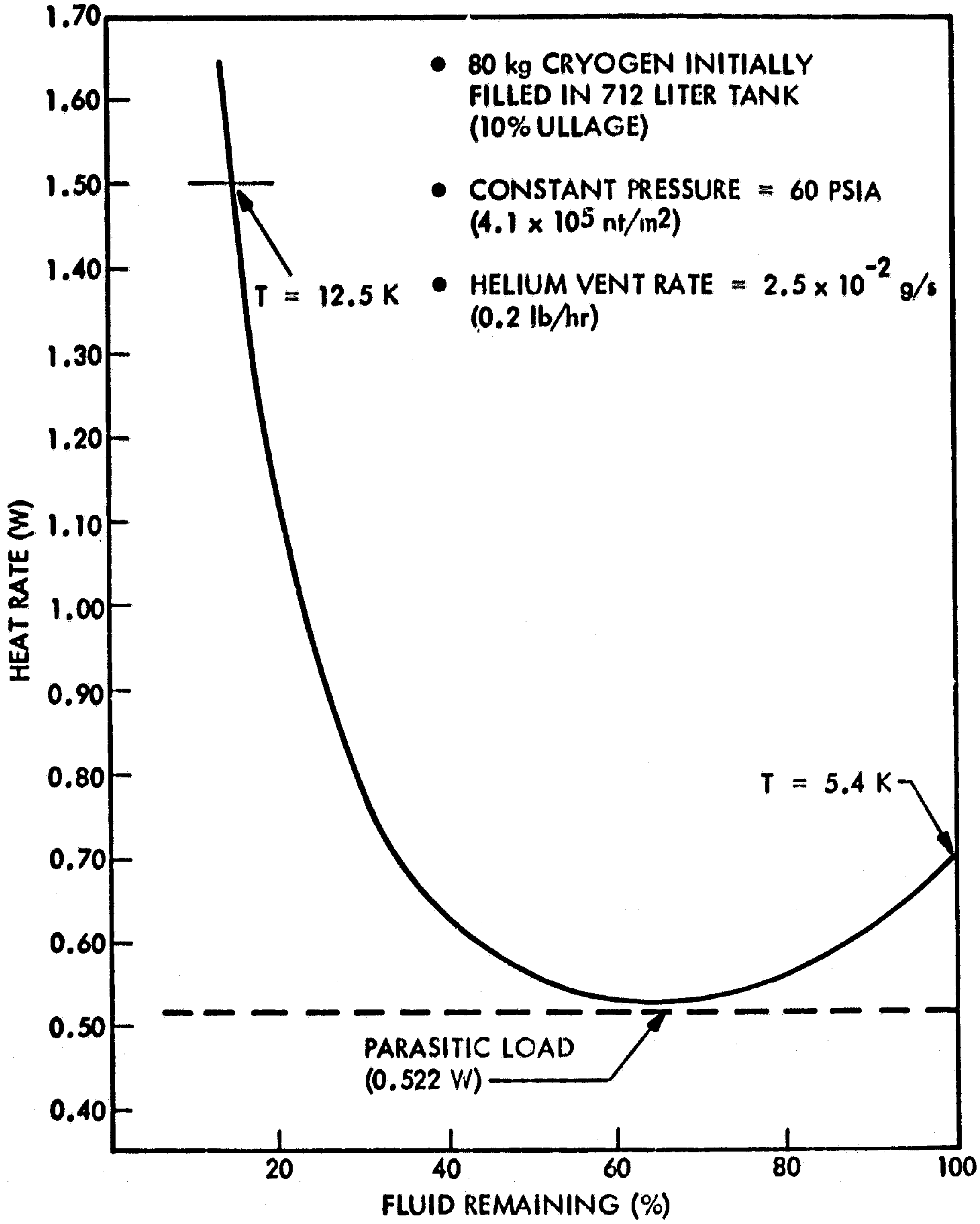
The relationship between the required heat input to the supercritical helium dewar for a constant vent rate of 2.5×10^{-2} gm/s(0.2 lb/hr) at 4.1×10^{5} nt/m^2 (60 psia) and the percentage of helium remaining in the dewar is shown in Fig. 4-6. To insure no more helium is vented than required, the parasitic heat load to the tank must be below the minimum required heat rate of 537 mw which occurs at the 65% filled condition.

A nodal thermal model of the CLIR cooler was constructed to determine cooler heat rates which includes both radiation and conduction coupled heat transfer effects. Computations were made using the Lockheed developed thermal analyzer program, THERM. THERM can be used to solve both transient and steady state temperature and flow problems by use of a forward finite difference algorithm for solving an analagous R-C electrical network.

Detail between the ambient and vapor cooled shield portion of the cooler were not included in the nodal analysis. Rather, cooler heat rates were evaluated solely as a function of the vapor cooled shield temperature which was varied in each configuration from between 100°K to 150°K.

A total of 78 nodes were used in the analysis of which 67 described sections of the multilayer insulation. Each MLI node between the vapor cooled shield and the cooler or support tube is 0.397 cm thick with a total of 8 node thicknesses between the shield and cooler. Width of the nodes over the cooler were selected to give three equal surface area sections. Between the support tube and vapor cooled shield, the widths were selected to give six equally spaced node sections. The section nearest the cooler contains 7 1/2 node thicknesses and because of the slant angle between the support tube and vapor cooled shield, each succeeding section contains one less node thickness until the final section, which contains 2 1/2 node thicknesses. In all sections the 1/2 thickness occurs because of the wedge formed by the slant angle between the support tube and node, which is assumed parallel to the vapor cooled shield.

FIG. 4-6 REQUIRED HEAT INPUT DURING HELIUM WITHDRAWAL



Both parallel and perpendicular heat transfer effects between adjacent MLI nodes was included. In addition, the conductivity of both the fiberglas support tube and MLI (parallel and perpendicular) were assumed to be temperature dependent.[3,4,5] Emissivity of the cold experiment surfaces was assumed to be 0.1, a conservatively high value.

Five different configurations were considered, each assuming the fiberglas support tube thickness to be 0.203 cm. Varied was the type of boundary condition imposed on the end MLI nodes - either thermally shorted or radiatively coupled to the adjacent fiberglas support tube node - as well as the use of a 1000 Å gold coating along the outer surface of the support tube. Also considered was the use of a sandwich construction fiberglas support tube as discussed in Section 4.5. The sandwich support tube is also 0.203 cm thick having a 0.025 cm thick inner and outer fiberglas wall over a 0.152 cm thick center of epoxy or foam. The configurations considered are summarized in Table 4-1 and the results are shown in Fig. 4-7. Also shown are the one-dimensional hand calculation performed for cases 3 and 5. As can be seen, good agreement to within 10% is achieved at all shield temperatures, indicating a small effect from parallel heat flow in the MLI.

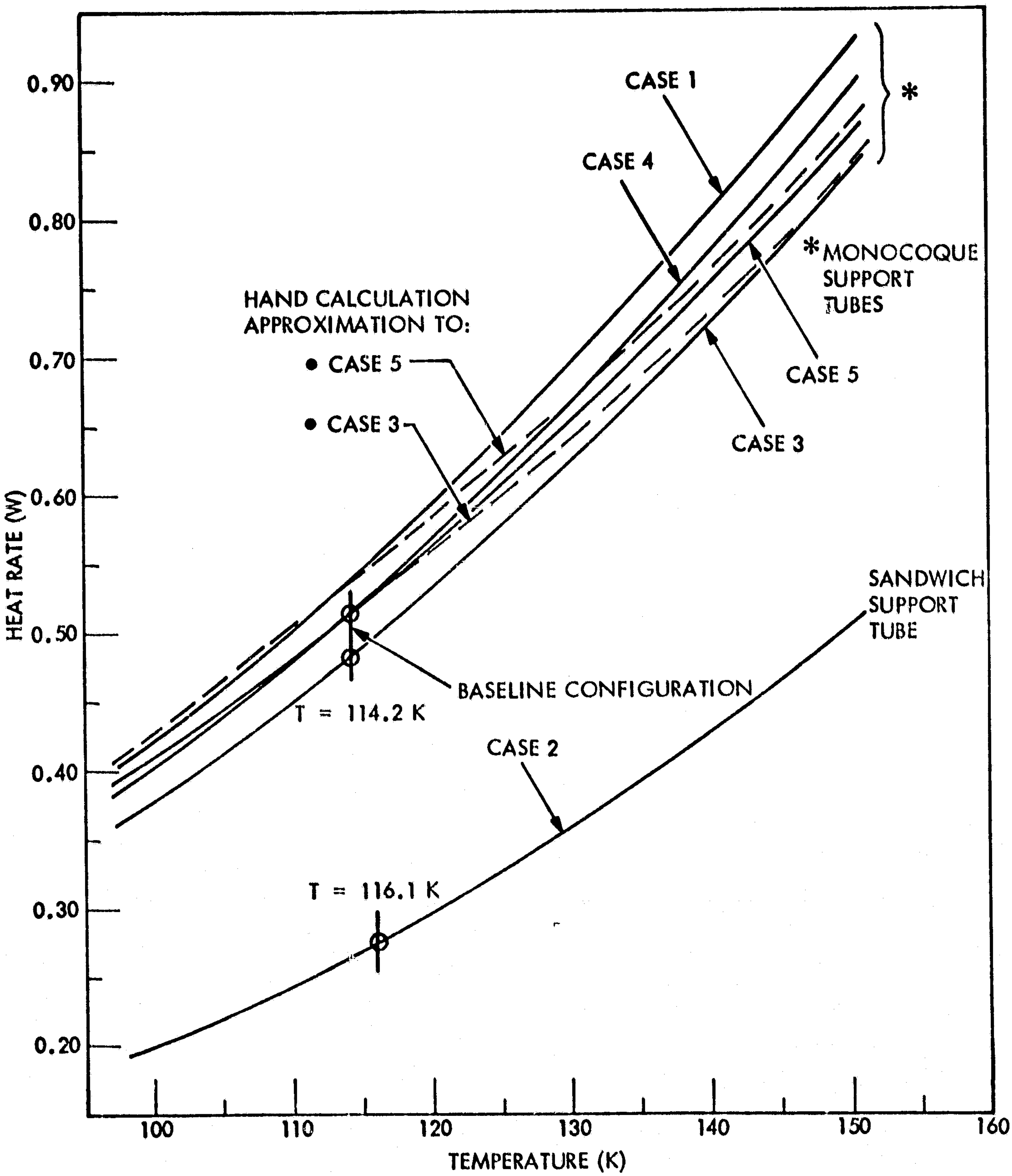
The conduction heat leak from the gold coating overwhelms any benefit gained by reducing the radiation coupling to the MLI edges and will not be used on the cooler. Cases 3 and 4 bracket the dewar heat load lower and upper bounds respectively.

To determine the shield temperature, a one dimensional heat transfer analysis is performed equating the net parasitic heat load to that removed by the incoming cold vent gas at 104°K for several shield temperatures between 100 and 150°K. In calculating the net heat load, the heat removed from the shield to the dewar as found from the nodal analysis is included. Results show that the change in shield temperature caused by the difference in the net heat load change between cases 3 and 4 causes a negligible (0.2°K) change in the vapor cooled shield temperature which will run at 114.2°K. At this temperature, the parasitic heat

TABLE 4-I CONFIGURATIONS STUDIED IN THERMAL ANALYSIS

CASE	SUPPORT TUBE		GOLD COATING		MLI BOUNDARY COUPLING TO SUPPORT TUBE	
	MONOCOQUE	SANDWICH	YES	NO	RADIATION	THERMAL SHORT
1	X		X		X	
2		X	X		X	
3	X			X		X
4	X			X	X	
5	X		X			X

FIG. 4-7 THERMAL ANALYSIS RESULTS



load to the dewar will be between 488 to 522mw depending on the MLI boundary, less than the required minimum of 537 mw to provide venting at a rate of 2.5×10^{2} gm/sec. To maintain this flow rate constant, between 15 mw and 991 mw (@ 12°K) auxiliary heating will be required during the mission.

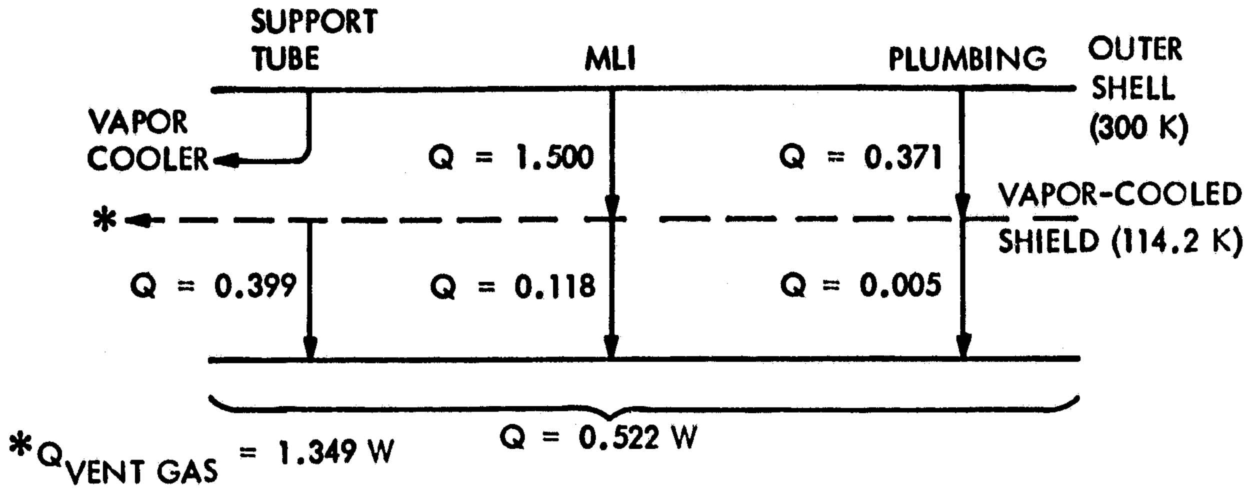
A breakdown of the cooler heat loads for the case of configuration 4 is shown in Fig. 4-8 at the 2.5×10^{2} gm/sec flow conditions. A sensitivity analysis on the vapor cooled shield shows that a cooler heat rate change of 2% will occur for every 10% change in the shield heat rate. Note that no support tube heat leak exists into the vapor cooled shield because of continuous vent gas cooling utilized along the lengths of this tube section. The major heat leak is through the fiberglas support tube - 76% of the total 522 mw.

Realizing that the 522 mw heat leak is marginally close to the required 537 mw, the concept of a sandwich construction support tube was considered in case 2. Because of a lower heat leak to the dewar the shield will increase in temperature slightly to 116.1°K at which point the heat leak to the dewar will be 282 mw, - a 46% reduction.

Although the present baseline design provides for a workable system, the much more comfortable safety margin which can be achieved by implementation of the sandwich construction support tube drives the final design towards incorporation of the sandwich construction support tube. Before this can be done however, sandwich tubes need to be constructed and subjected to appropriate testing to verify confidence in the concept.

The selection of a 0.203 cm thick support tube was made as a result of some of the early structural analyses. Later, more refined analyses which were completed after the thermal design studies were performed indicate that the 0.203 thick tube is overdesigned and that a 0.152 cm thick tube meets the structural criteria. Expected cooler heat rate using a 0.152 cm thick support tube would drop from 0.522 watts to 0.434 watts -- a figure more in line with the heat rates predicted in the phase I cooler design, 0.415 watts.

FIG. 4-8 SUMMARY OF THERMAL CONDITIONS



	CONFIGURATION	HEAT LOAD TO CRYOGEN (W)
SUPPORT TUBES	0.203 cm-THICK RIGHT CIRCULAR TRUNCATED CONE 73.6 cm SLANT LENGTH 65.9/57.7 cm MAXIMUM/MINIMUM RADIUS	0.399
MULTILAYER INSULATION	OVERALL THICKNESS = 6.35 cm VAPOR COOLED SHIELD LOCATED 3.05 cm FROM CRYOGEN TANK SILK NET/DOUBLE-ALUMINIZED MYLAR $K = \begin{cases} 5.3 \times 10^{-7}\ \mathrm{W/cm-K}\ (114.2-300) \\ 5.8 \times 10^{-8}\ \mathrm{W/cm-K}\ (6-114.2) \end{cases}$	0.118
PLUMBING	STAINLESS STEEL 1.27 – cm DIAMETER 0.013 – cm WALL THICKNESS	0.005

4.5 STRUCTURAL ANALYSIS

The BOSOR4[6] computer code was used to make a structural analysis of the CLIR cooler. The model used in the analysis, as drawn by the computer system, is shown in Fig. 4-9. The mass of the experiment is represented by a cylinder with uniformly distributed mass (310 lbs), and the mass of the cryogen gas and tank is represented by an ellipsoid (274 lbs), also with a uniformly distributed mass. The experiment is assumed to be rigidly supported to the "strongback" since the details of the support were not available. Additional analysis will be required when the mounting technique is established. The structural parts are all modeled as shells with their masses distributed according to the thickness distribution. The figure represents the mean surfaces of all the items making up the model.

The major structural items are:

- The vacuum shell, made of 0.15 inch thick aluminum. The conical part is stiffened by 3 rings, each 0.3 inch wide and 0.5 inch high. The bottom ellipsoidal part is 2 inch thick (waffled configuration) at the support point (at radius = 4 inch), gradually decreasing to 0.15 inch, at radius = 20 inch. The external collapse pressure of the vacuum shell is estimated to be about 30 psid.

- The support tube, made of 0.08 inch fiberglass/epoxy with the following properties:

Shear Modulus:	750,000 psi
Elastic Modulus, Axial:	5,550,000 psi
Elastic Modulus, Hoop:	2,190,000 psi
Axial Tensile Strength:	169 ksi
Axial Compressive Strength:	89.5 ksi
Hoop Tensile Strength:	28.6 ksi
Hoop Compressive Strength:	35.7 ksi

  The ends of the fiberglass tube are encapsulated in aluminum sleeves.

- The vapor shell, made of 0.03 inch thick aluminum.
- The cryogenic tank, made of 0.15 inch thick aluminum. (However, in the model shown in Fig. 4-9 the tank mass is included in the cryogen weight and its stiffness is practically infinite. A separate analysis of the tank was made, see below).

A nodal analysis and a dynamic stress analysis for lateral vibrations were made for the model shown in Fig. 4-9. The first frequency is 21.3 Hz, with a mode shape as shown in Fig. 4-10. This mode involves primarily the vacuum shell and the experiment package. The second mode, 59.4 Hz, see Fig. 4-11, is the most significant one for the fiberglass support tube, contributing more than 90% to the random vibration stress. Higher modes, up to the sixth, are shown in Figures 4-12 through 4-15.

The static and dynamic environment is described as follows:

Static acceleration, ultimate:	4.95 g
Sinusoidal vibration:	0.5 g, 5-40 Hz
	0.77 in/sec, 40-80 Hz
	1.0 g, 80-200 Hz
	+6 dB/Oct, 20-130 Hz
	0.18 g^2/Hz, 130-1000 Hz

The sinusoidal vibration refers to the lateral vibrations, which is the critical case and the only one investigated here. No sinusoidal vibration requirements are given for more than 200 Hz.

The maximum axial support tube <u>stresses</u> occurs at the narrow end, and are:

Static Load:	0.4 ksi
Sine vibration, 1st mode:	0.2 ksi
2nd mode:	0.2 ksi
3rd mode:	0.1 ksi
Random, 3σ (99.7%):	2.2 ksi

The damping factor used in the above analysis is 2.5%. We note that the stresses listed above are only a few per cent of the strength of the tube.

At the 99% design level, the buckling allowable of the support tube is 129 lb/in axial force. The maximum applied axial force is the 3σ random vibration force, which is 117 lb/in. Thus, the probability of buckling of the tube in random vibration is less than 0.00003. The applied load in the sine vibration case is 65.5 lb /in, giving a margin of safety of 1.8 at the 99% design level.

Thus, the 0.08 inch thick tube is quite adequate to its task, and could probably be made somewhat thinner, perhaps 0.06 inch. Even less tube material would be required if the tube is made in sandwich construction. A possible design would be two glass layers of facing, each 0.02 inch thick, separated by a 0.1 inch thick layer of epoxy. This results in a wall design equivalent to about 0.05 inch thick from the thermal point of view, and more than 0.1 inch thick from the structural point of view.

The relatively low first frequency, 21.3 Hz, is due to the very narrow base support assumed for the ASPS (only 4 inch radius, see Fig. 4-9. An improvement would be to expand the support to, say, a radius of at least 20 inches, or to provide a secondary support point at that point. If the payload were "tipped over" and latched during launch the first mode frequency would be substantially increased. This study should be performed when the instrument and pointing system are more fully defined.

The cryogenic tank, with a wall thickness of 0.15 inch, experiences a total axial growth of 0.105 inch when an internal pressure of 60 psid is applied, see Fig. 4-16. The classical buckling pressure is 138 psid, external, with the buckle appearing at the pole. The structure is imperfection-sensitive, so a "knock-down" factor of 1/4 is appropriate, leaving a design buckling pressure of 34.5 psid, which is quite adequate, giving a safety margin of 2.3.

FIG. 4-9 INITIAL UNDEFORMED STRUCTURE

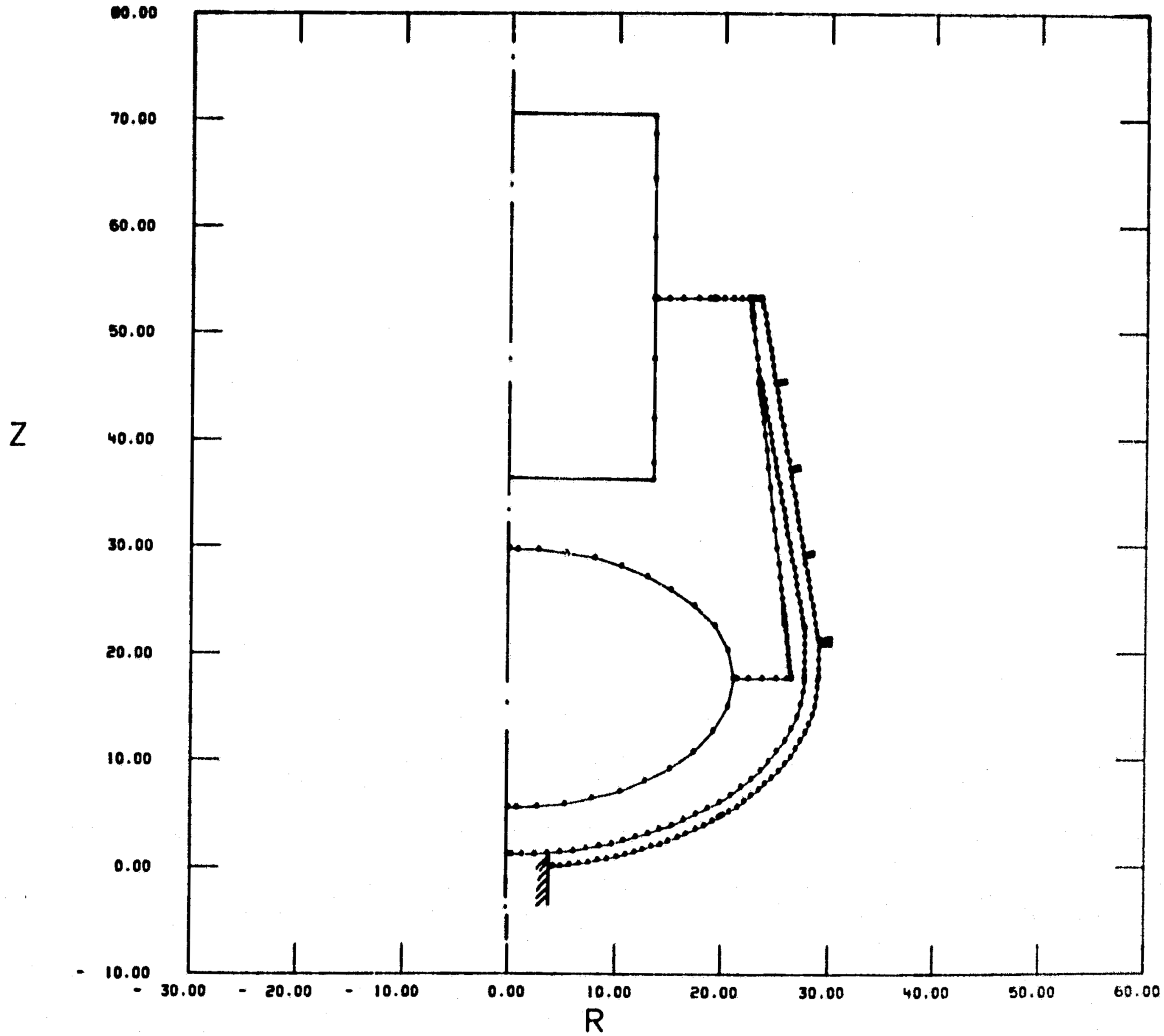


FIG. 4-10 DEFORMED STRUCTURE MODE NO. 1 2.127+01 CPS.

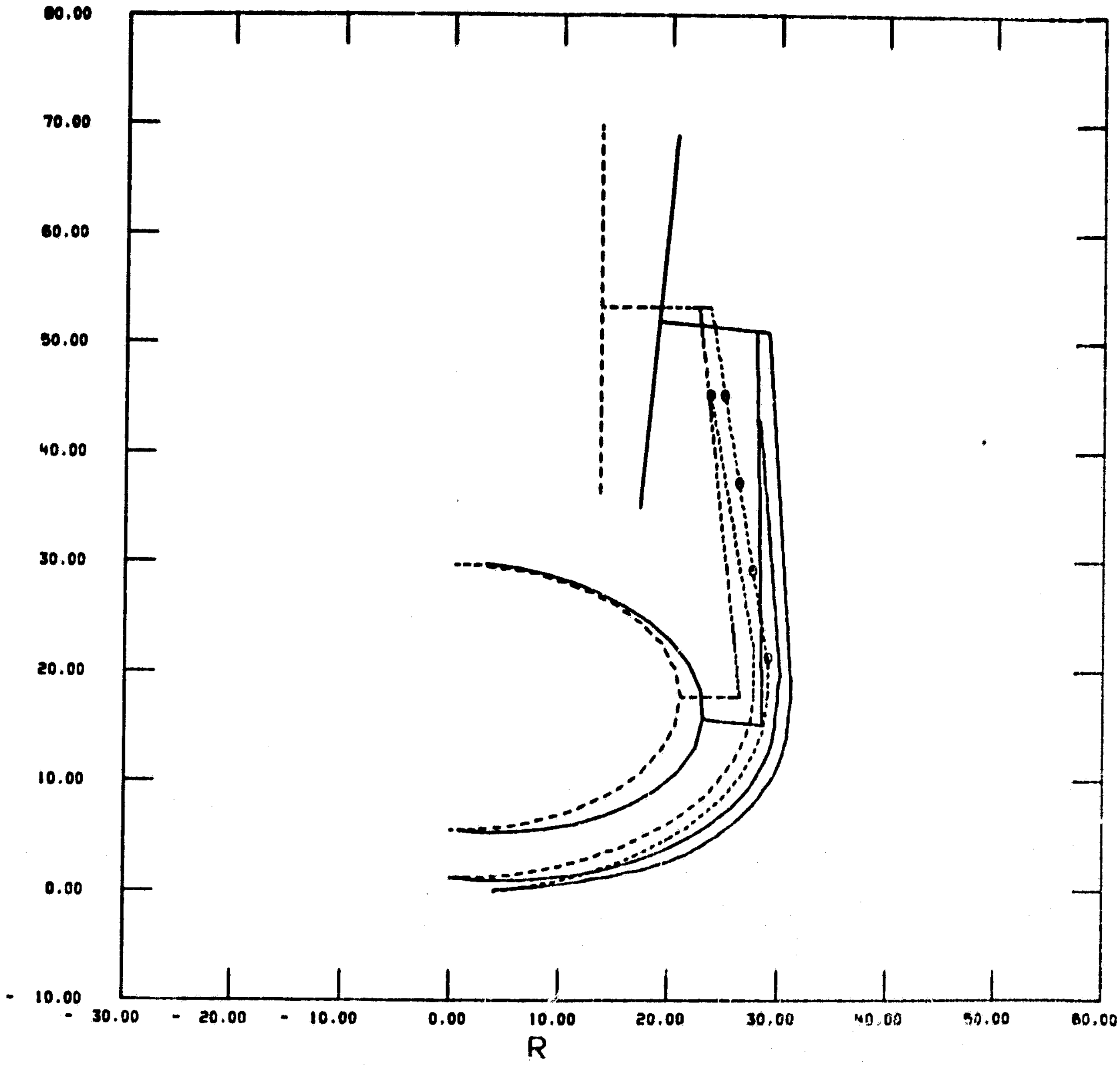


FIG. 4-11 DEFORMED STRUCTURE MODE NO. 2 5.942+01 CPS.

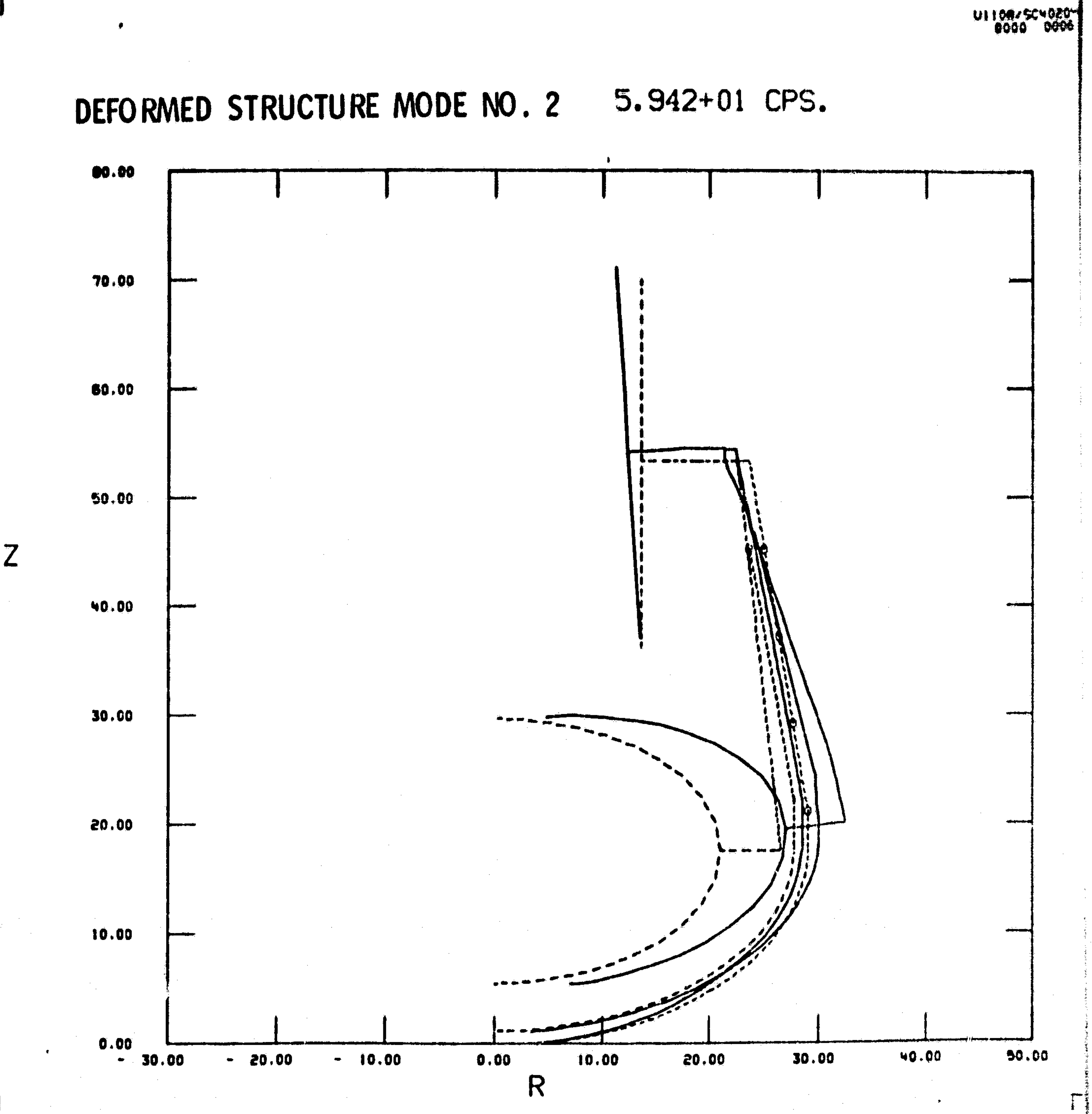


FIG. 4-12 DEFORMED STRUCTURE MODE NO. 3 2.013+02 CPS.

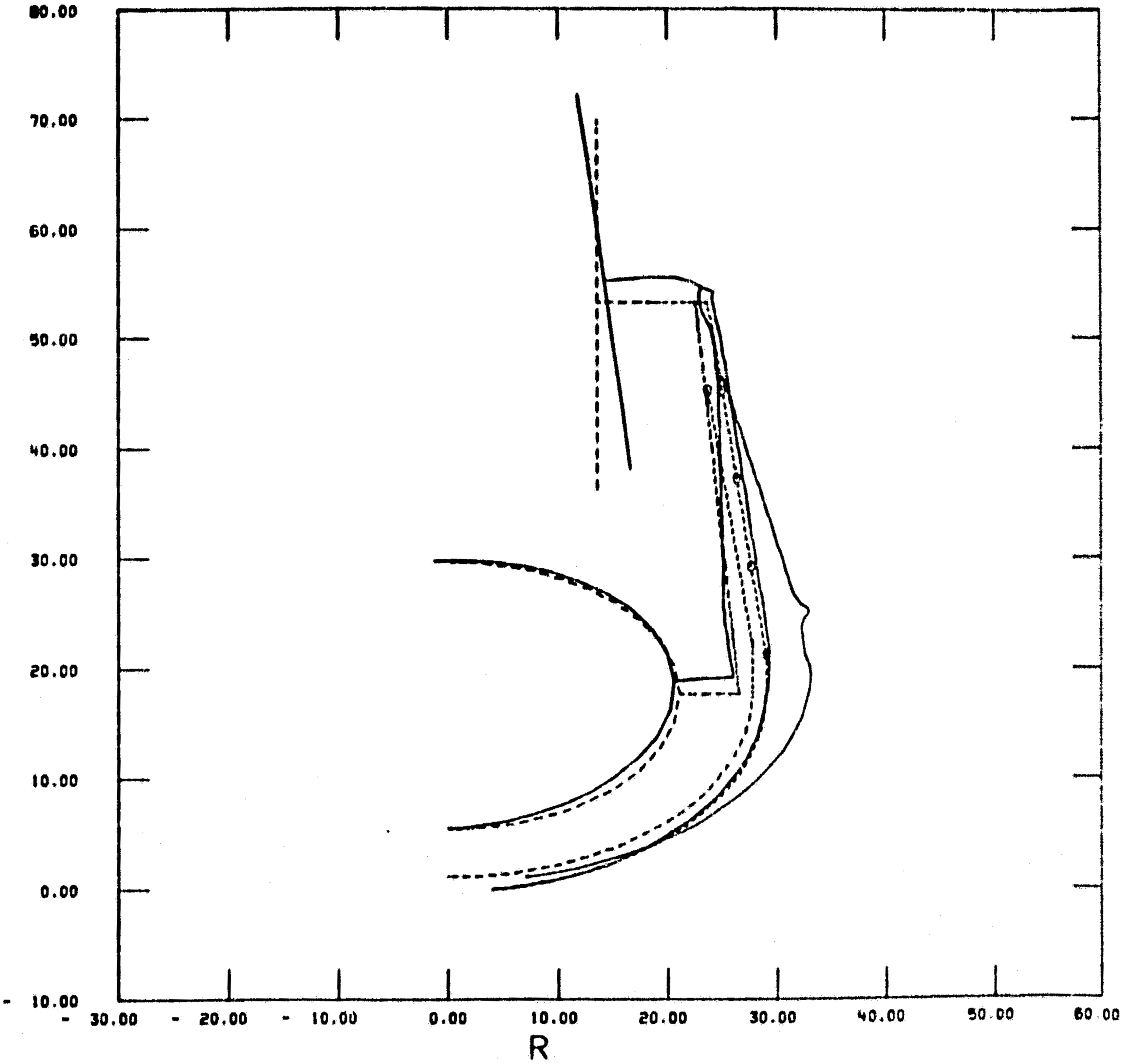


FIG. 4-13 DEFORMED STRUCTURE MODE NO. 4 2.762+02 CPS.

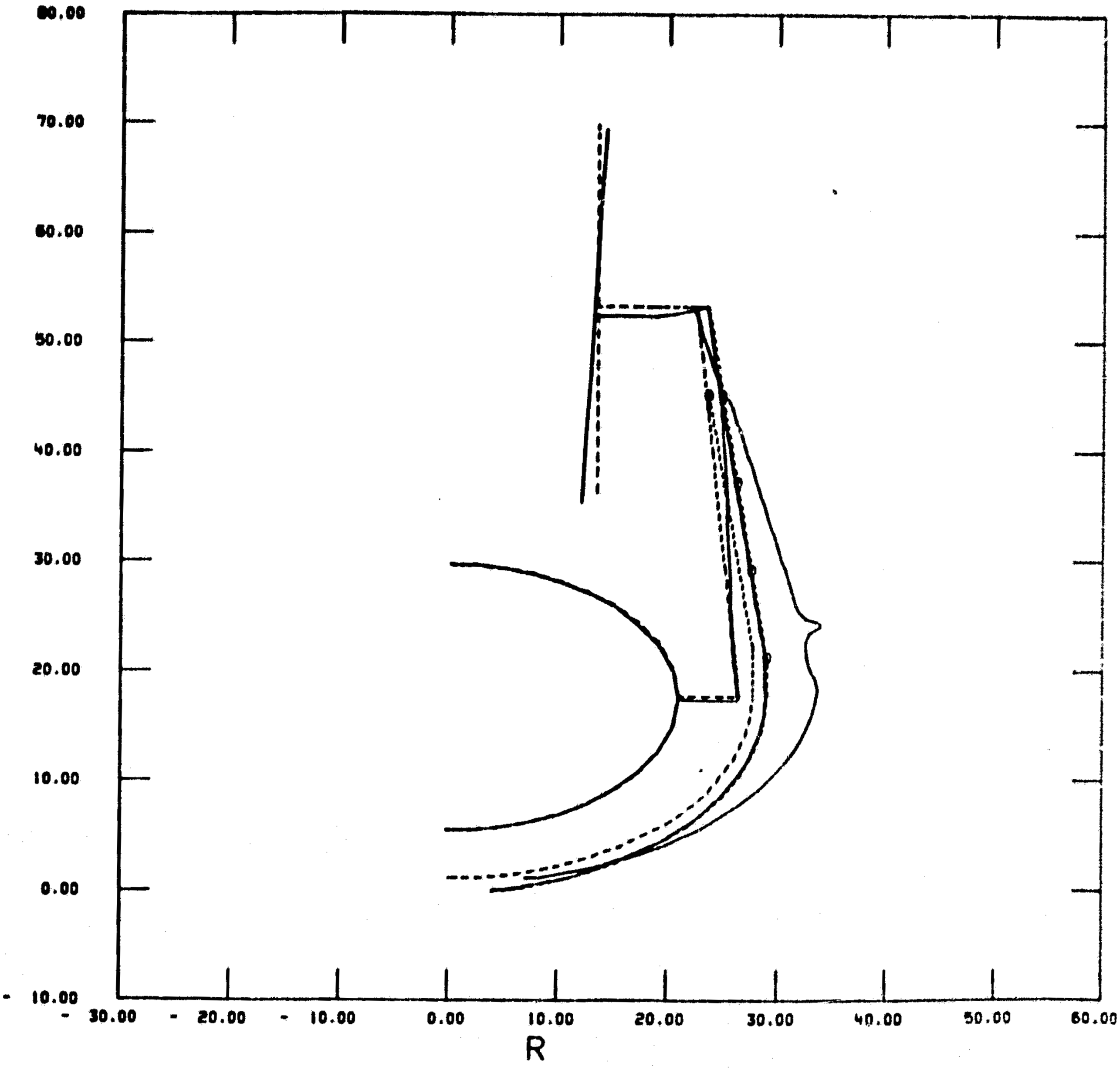


FIG. 4-14 DEFORMED STRUCTURE MODE NO. 5 3.474+02 CPS.

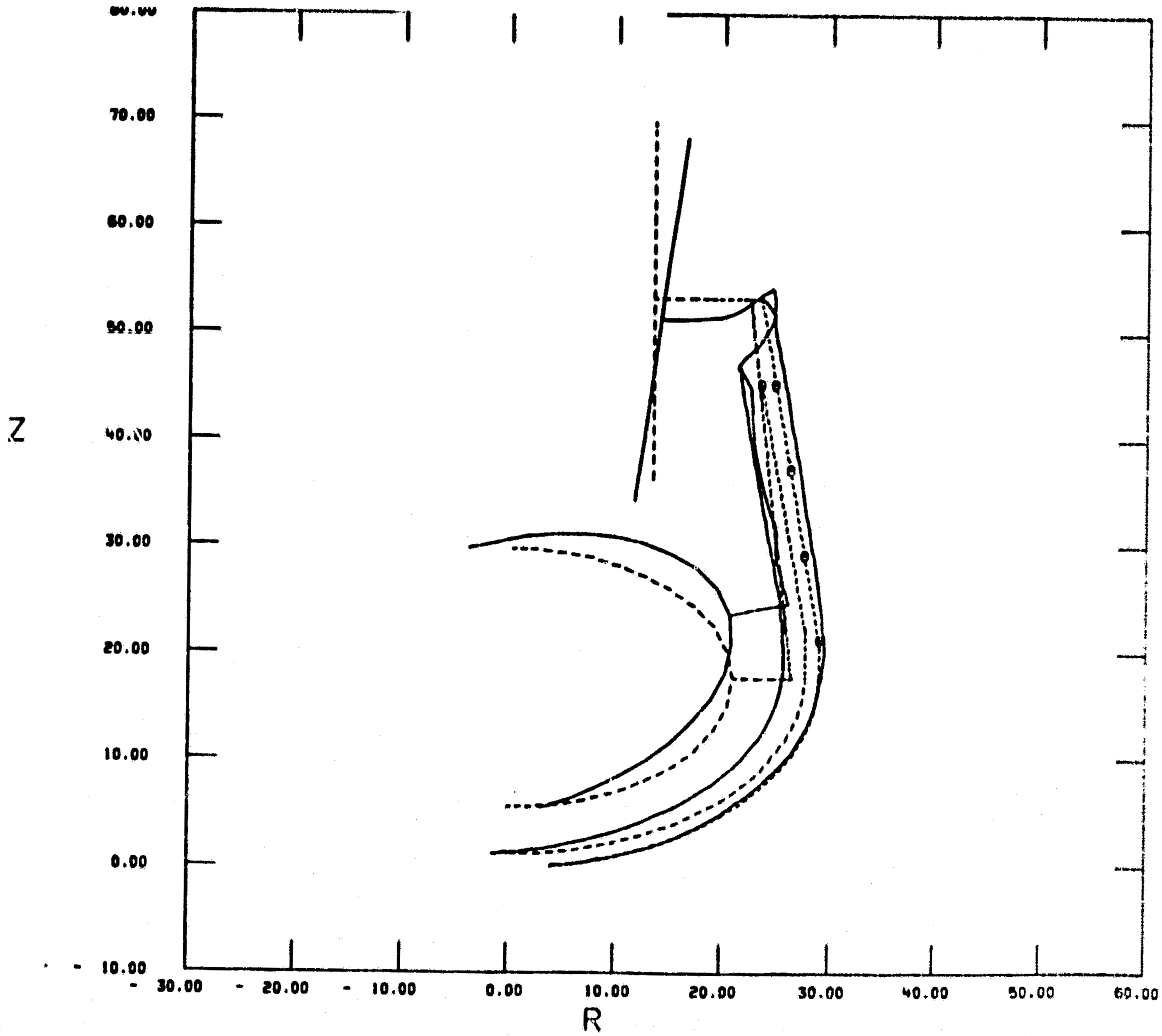


FIG. 4-15 DEFORMED STRUCTURE MODE NO. 6 5.481+02 CPS.

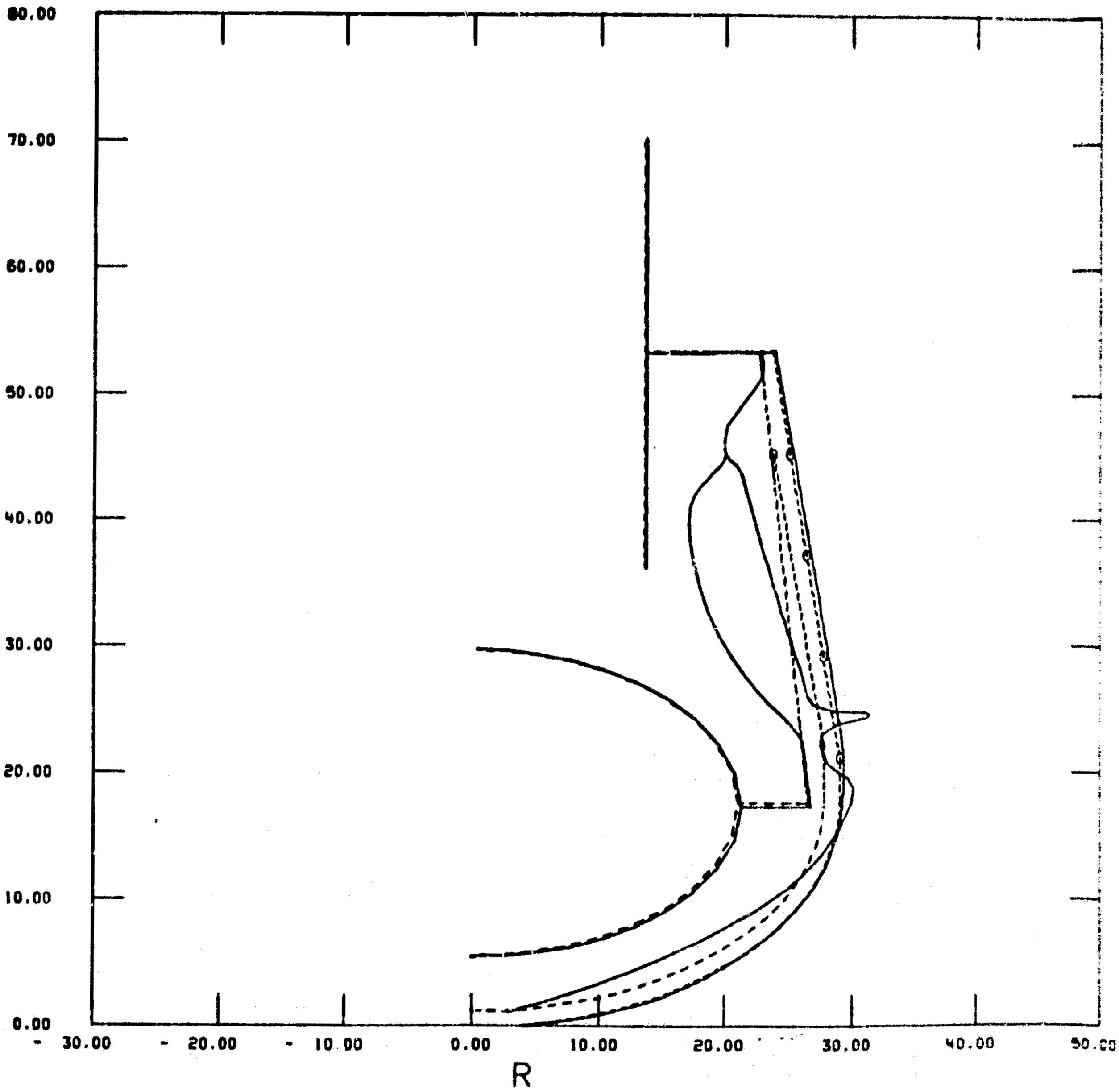
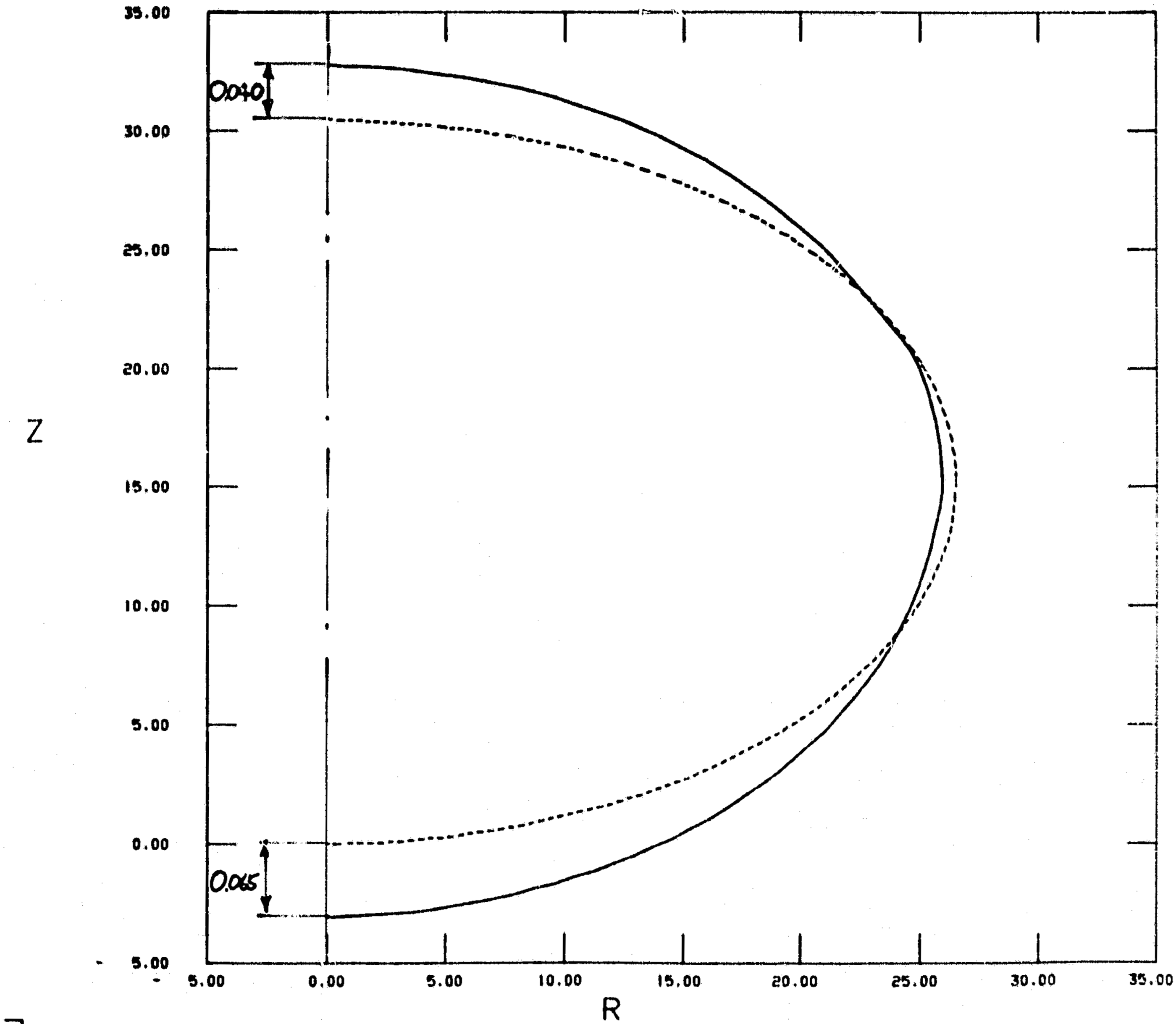


FIG. 4-16 CRYOGENIC TANK DEFORMED STRUCTURE

U1108/SC4C. 0000

LOAD STEP 3, LOAD= 6.000+01 PRESTRESS



5.0 SUMMARY AND RECOMMENDATIONS

This Phase II Study has resulted in the design of a cryogenic system to be utilized with an end mounted pointing system (ASPS). The analysis indicates that the resulting design is significantly heavier than the design which evolved during the Phase I Study for the C.G. mounted pointing system (SIPS). This additional weight (340 Kg vs 197 Kg) for Phase I design is largely due to the increased diameter of the system resulting in greater tank and vacuum enclosure weights. Additional weight is also required in the area where the payload is attached to the pointing system in order to provide additional stiffness to the payload. The vacuum shell required to enclose the instrument is shorter in this design and therefore an attendent reduction in instrument package weight would be expected.

Although the cryogenic system design appears to allow a straightforward integration of the instrument package, many areas of the instrument-cooler interface remain to be defined before compatibility is assured. The heat rates to various regions of the instrument are dependent upon the cooler design and may change when a coupled thermal analysis of the instrument and cooler is performed affecting the cryogen requirement.

The primary resonance of the cooler-instrument package occurs at 21 Hz. This mode is one in which the flexibility of the vacuum shell allows motion of the cryogen tank and instrument. This resonance can be increased by additional stiffening in the region of the pointing system-vacuum shell connection at the cost of additional weight.

The second resonance at 59 Hz is due to vibration of the helium tank on the fiberglas support tubes.

In the thermal area the predicted heat loads to the system (0.522W) are slightly below the values required for constant pressure expulsion of the cryogen (0.55W), however, it is felt that an additional margin is desirable to assure satisfactory system operation.

The parasitic heat load can be reduced to 0.28W by the utilization of a sandwich support tube structure in place of the monocoque structure assumed in the design. This structure has not been utilized for cryogenic support to the writer's knowledge and therefore would require some development testing before incorporation into the design. It is recommended that this development be pursued during the detail design phase of the cooler and that both support configurations be studied further before a final selection is made.

The cooler design does not appear to require any substantial new technology development (except perhaps the sandwich support tubes) and appears to be relatively straightforward.

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